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SPUR GEARING.

Paper presented to the Institution, London Section, 5th December, 1929, by G. R. Marsh (Member).

THE first types of spur gearing put into general use were made either with cast teeth or centres of cast iron and wooden teeth, usually hornbeam.

The workmen who fitted these teeth took a great deal of pride in shaping them, the work being performed by hand to a template. The advantage of this form of gearing was that in the event of one or more teeth breaking out, fresh ones could be inserted without scrapping the whole gear. The construction was, however, only applicable where the pitch line speeds were low. With the increasing demand for gears to meet higher speeds it was necessary to introduce mechanical methods for machining gear teeth of an accuracy corresponding to the speeds required. The cycloidal form of tooth was almost universally adopted by gear manufacturers, the addendum or portion above the pitch line being formed by a point on a circle rolling outside a cylinder and the dedendum being formed by a point on a circle rolling inside the cylinder. Machining methods were based on a forming process so that the cutters or tools were shaped before hardening to give the required

profile. Two serious defects in the cycloidal form of profile have resulted in it being almost completely discarded. The first defect was that this form of tooth would only work correctly if the pinion and wheel were maintained exactly at a pre-determined centre distance, any variation in this centre distance causing serious interference with the correct meshing. The second defect was that this profile did not lend itself to any simple method of generation and its production was consequently restricted to form cutter processes. The involute tooth profile which has now been almost universally adopted as standard has a single curve which is produced by a point on the circumference of a disc as it is unrolled along a straight line. The shape of this curve is automatically fixed immediately the centre distance, ratio and pressure angle for any pair of gears are decided. The base circle from which this involute is generated is one of the most important dimensions in a gear. An increase in centre distance will result in the pitch circles being altered, but the base circles will remain the same and the total effect of this increase in centres is to alter the pressure angle at which the teeth run. There will be no interference whatever with the meshing conditions or the continuity of the motion transmitted providing the centres are not increased to such an extent that the duration of contact is less than the pitch. This duration of contact varies with the number of teeth in the pinion and wheel, and their shape. Further consideration will also show that an involute has no definite pressure angle until some point on it has been fixed.

There are two well-known methods of designing teeth to avoid undercut, and to improve the conditions of engagement. The pressure angle at which the gears are to run may be increased or, alternatively, the addendum of the pinion may be increased and that of the wheel decreased by a corresponding amount. Either modification will reduce the distance between the root circle and base circle and so diminish the undercut. Correction on the pinion teeth can only be made at the expense of the wheel teeth and this method is, therefore, most useful where there is considerable difference between the number of teeth in the pinion and wheel.

In the case of a pinion and wheel having 10/60 teeth and $14\frac{1}{2}°$ pressure angle, there will be considerable undercut and only a portion of the whole tooth profile will be used for engagement. If the pressure angle is increased to 20°, there will still be a considerable amount of undercut on the pinion, but the tooth form is appreciably better than with a $14\frac{1}{2}°$ angle and, in addition, we now have the duration of contact more than one pitch, whereas, in the first case, it was less. It is of utmost importance that the duration of contact is greater than one pitch to secure continuity of motion.

If the line of action is drawn, it will be easy to determine the

points on the pinion and wheel teeth where contact takes place at any moment. Contact between the teeth will normally finish where the line of action joins the base circle or where the outside diameter of the wheel cuts the line of action, whichever occurs first. If undercut extends above the base circle, this defines the duration of contact. By increasing the pressure angle to 30° the whole of the wheel tooth is brought into service, the main objection to such a large pressure angle being that the pinion teeth become very pointed and the load on the bearings is also slightly increased.

Another method of avoiding undercut is by increasing the addendum of the pinion and decreasing that of the wheel by a corresponding amount.

As previously mentioned, this form of correction is most useful where there is a considerable difference between the number of teeth in the pinion and wheel, and becomes less effective as these numbers become equal. Its advantages are that it brings the root radius of the gear nearer to the base circle and, consequently, reduces undercut. As a direct result, the base of the tooth is stronger. In addition, this form of correction in many cases enables $14\frac{1}{2}°$ cutters to be used to give a pinion which has no appreciable undercut, where otherwise a 20° cutter would have to be employed giving a pressure angle larger than necessary. It will generally be found that the smaller the pressure angle adopted consistent with strength and a correct ratio between duration of contact and pitch, the more quietly the gears will run, and this is of particular importance at high speeds.

In many cases the best results will be obtained by combining a modified pressure angle with an addendum and dedendum correction.

A system has been developed by the Maag Company of Zurich by means of which the best form of tooth for any gear ratio can be immediately calculated and set out without resorting to "Trial and error" methods. It is not proposed to discuss the method adopted, but in general, the idea is to use a range of cutters all having 15° pressure angle to produce gears having varying pressure angles. The corrections embodied are, variation in pressure angle, addendum and dedendum, and tooth height. The system has the further advantage that any ratio can be fitted into a pre-determined centre distance with the correct amount of backlash, securing excellent meshing conditions.

While on the question of undercut, it will probably be of interest to show how the exact amount can be determined and also how a generated profile may be completed for any given form of cutter, Fig. 7. The heavy lines show a rack form of cutter with its centre line and the pitch circle of the gear.

The line of action, E.E. has been filled in. Points 1 to 10 have been marked off approximately equi-distant and lines carried off

at right angles to the cutter profile to meet the pitch line of the cutter at 1A, 2A, etc. Lines are also carried off from points 1, 2, 3, etc., parallel to the pitch line of the cutter until they reach the line of action, 1B, 2B, etc. From points above the base circle with centre "O," arcs are carried back to 6B to 10B.

Distances D1A, D2A, etc., are transferred to the pitch circle of the gear giving points 1C, 2C, etc. With distance 10 to 10A and centre 10C, an arc is described cutting the arc carried back from 10B at 10E. Similarly, with radius 9 to 9A and centre 9C arc from 9B is cut at 9E. In this way points for the involute are formed.

To find the exact shape of the tooth generated below the base circle, or what is usually called undercut, it is necessary to trace out the path of the tip of the cutter "K" as the pitch line of the cutter rolls round the pitch circle of the gear. This is shown by marking off from D points F. G. H., etc. These distances are then transferred to the pitch circle of the gear points F′, G′, H′, etc. With radius FK and centre F′, an arc is described. With radius GK and centre G′ another arc is described and so on. The curve formed where these arcs intersect gives the path traced out by the cutter tip. The point K of the cutter tooth is usually rounded off to form a fillet radius in the root of the tooth and an additional effect is to reduce the undercut and so strengthen the tooth section.

Tooth Strength.—When considering the strength of teeth for ordinary drives, it is advisable to assume that the whole of the load will have to be carried by one tooth and, moreover, that this load is applied at the tip of the tooth. In theory, there will usually be more than one tooth in engagement and even when one tooth is called upon to carry the whole load this will be applied some distance from the tip.

Considering the gear tooth as a cantilever

$$\text{Stress f} = \frac{\text{bending moment (P)} \times \text{moment of inertia of Section 1}}{\text{Distance of outermost layer from axis (Y)}}$$

p = tangential pressure × tooth height above the section where the maximum stress occurs.

$$1 = \frac{BD^3}{12} = 1 \times \text{thickness of tooth at point of maximum stress.}$$

$$Y = \frac{D}{2}$$

Material should be chosen so that stress, when allowing for the maximum overload, does not exceed one-fifth of the ultimate tensile strength of the material. A usual factor of safety is 8 to 10.

Where there are two teeth always in engagement and the methods of manufacture are such that the errors in pitch and shape are kept within very fine limits, the load can be assumed as carried equally on the two teeth and so smaller pitches may be adopted.

Of equal importance to tooth strength is the durability of the surface, and this depends on material, curvature of tooth profile, accuracy, pitch line speed and lubrication. These factors are so interdependent that it is very difficult to consider them separately;

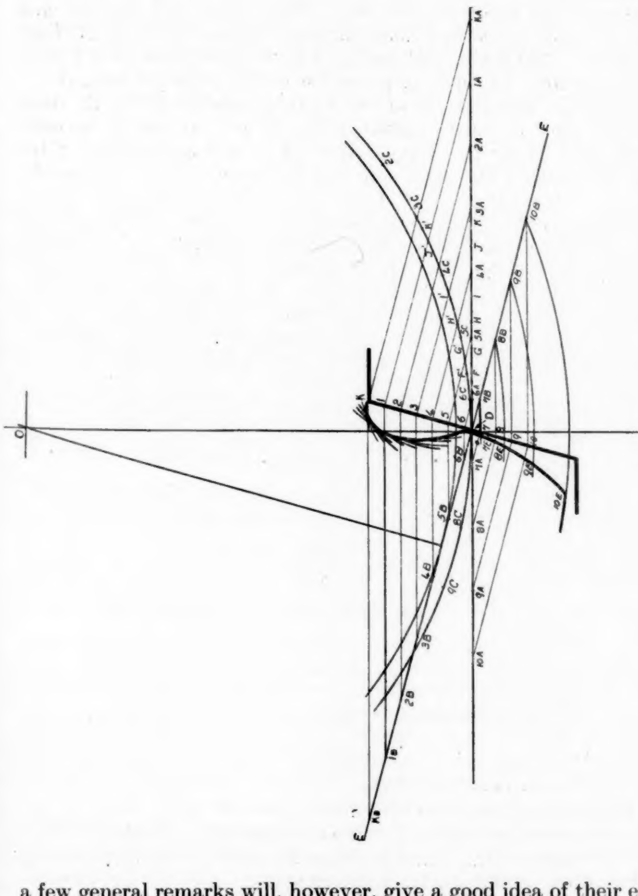

FIG. 7

a few general remarks will, however, give a good idea of their effect.

The choice of material must primarily depend upon the conditions to be fulfilled. For slow running drives, it is quite common to use a cast steel or forged steel pinion engaging with a cast-iron

wheel. In such a case, it is usually the strength of the cast-iron teeth which determines the maximum pressure permissible. If the power to be transmitted is considerable, a coarse pitch has to be employed which necessitates a fairly large pinion diameter in order to obtain a reasonable number of teeth. Where the conditions are severe or the space is restricted, a cast steel wheel is commonly employed, the higher tensile strength and toughness enabling smaller pitches to be adopted and so reducing the centre distances. Wherever conditions permit a forged steel wheel is preferable, the material being more homogeneous and of higher tensile strength.

Pinions of alloy or high carbon steels with the teeth case-hardened and ground are now widely used where there is a large ratio of reduction or where durability is important. In the case of a gear ratio of say 6 to 1 each pinion tooth passes through engagement six times as often as each wheel tooth and consequently should have proportionally higher wear resisting properties. Unless this provision is made it will be found that the pinion teeth wear more rapidly than the wheel teeth and in so doing lose their correct involute shape so that the transmission of motion is no longer uniform. These conditions tend to accelerate and decelerate suddenly either the pinion or wheel and the masses of these is such that shocks of considerable magnitude result.

Not only do hardened and ground pinions retain their correct shape but also have a most beneficial effect on the relatively soft wheel teeth, acting as burnishers and rolling out the small inaccuracies left from the cutting operation.

For high speed work, particularly in the case of turbine reduction gears and similar drives, case-hardening steel is used for both the pinion and the wheel. Provided a suitable material is selected, we get after the heat treatment a glassy hard surface for resisting wear and a very tough core to carry the load. The following specification shows the type of steel generally used for this class of work.

Analysis.

C	0.10 to 0.20%.
Mn	Not over 0.50%.
S. & P.	,, ,, 0.05%.
Cr	,, ,, 1.00%.
Ni	3.00 to 3.75%.

Test after Treatment.

Tensile strength	50 to 70 tons per square inch.
Yield point	not less than 40 tons per square inch.
Elongation	15 to 12%.
Reduction of area	35 to 30%.
Izod impact	30 ft.-lbs.

Unfortunately, case-hardening steels distort during the heat

treatment and it would be impossible to use such gears for high speed work without first removing this distortion. This operation, however, can be fairly easily performed to-day and one process of gear grinding will be described later. Under favourable conditions these case-hardened and ground gears may be loaded up to 1,200 lbs. pressure per inch width at pitch line speeds up to 12,000 ft. per minute, and it is quite common practice to work on pressures of 800-lbs. per inch width at this speed.

Curvature of the tooth profile has some effect on its load carrying capacity as the tooth surfaces must flatten at the point of contact until stress introduced into the material balances the pressure. The profile has its maximum curvature near the base circle and gradually flattens out as it develops. The best working profiles therefore, are obtained when the base circle falls beneath the root circle.

Speed and accuracy are so closely related that it is proposed to deal with them together. As the speed of a gear increases, the most important factor is that changes in angular velocity have to take place in a shorter space of time. Such changes are caused by inaccuracies in the pitch or profile of the teeth and result in shocks, varying with the magnitude of the errors. In an example given later it will be shown that at high speed, these shocks assume serious proportions.

As the gear is speeded up, there is an increase in the rate of slide of the teeth relative to each other, but this does not seem to have any detrimental effect providing adequate lubrication is maintained.

There is a change of direction of slide at the pitch line and if the oil film were to break down it should take place at this point. At the pitch line, however, there is pure rolling and as the contact passes down the tooth profile the rate of slide increases and an oil film is again built up.

It is somewhat difficult to define what limits of inaccuracy are permissible for various speeds as, apart from all other factors noise which is permissible in one installation is quite intolerable in another. It is a simple matter to determine exactly what tone the gear will emit when running, as with a certain number of teeth passing through engagement per minute, there is a similar number of vibrations which give a distinct note. By altering the number of teeth, it is possible to move this tone higher or lower in the scale as required. The most objectionable period is between 550 and 4,000 pulsations per second and this is, unfortunately, a range into which many gears fall. The intensity of this note, ho'vever, is determined by the magnitude of the vibration which, in turn, depends upon the extent of the errors.

In a graph prepared by Mr. Daniel Adamson for his Paper, read before the Mechanical Engineers in 1916, he gives the relation

between accuracy and speed. It may be interesting to know that the permissible inaccuracies for ground gearing at 10,000/15,000 ft. per minute form an exact continuation of his curve.

Turbine generator sets of the small and medium sizes with the turbine developing anything up to 10,000 h.p. at speeds between 3,000 and 10,000 r.p.m. present interesting problems of the relation between speed and accuracy.

Take as an example a drive for transmitting 2,000 h.p. and increasing speed from a turbine running at 2,980 r.p.m. to a rotary compressor running at 4,130 r.p.m. the pinion having 70 teeth and the wheel 97. In such a set of gears an error of 0.0004″ extending one-third of the height of the profile would give a theoretical increase in tooth loading of 3,500 lbs. per inch, or 7.8 times the normal load of 460 lbs./inch. Although the actual increase in load is appreciably reduced, due to the buffering effect of the oil film and to the elasticity of the material at the surface of the gear tooth, the shock load would, nevertheless, be very considerable and for high speed gears of this type, the tooth to tooth errors should not exceed 0.0002″.

It is quite certain that all teeth deflect under load, the amount of this deflection depending on pressure, material and the distance of the point of contact from the root of the tooth when the maximum bending conditions are experienced.

The relation of duration of contact to pitch is usually about 1.25 to 1 and this means that the tooth immediately in front of the one about to enter engagement is making contact a little below the pitch line. In this position, both the driving and driven teeth are in a good position to resist bending, and interference when the new teeth enter engagement, is slight.

In high speed gears when there are several teeth in engagement the amount of deflection is so slight that correction can only be considered when the accuracy of the pitch and profile are of a very high order.

This correction at the tip of the tooth should, if possible, follow a parabolic form to give a uniform increase in velocity as the teeth enter engagement and should extend down the profile to the point where it is assumed that the tooth has taken up its share of the load.

Experiments are being conducted with a view to obtaining more definite data on this intricate subject, but the fact that there are large numbers of drives transmitting up to 10,000 h.p. and running at speeds up to 15,000 ft. per minute which have been in operation for years without any signs of fracture, proves this correction to be something in the nature of a super-refinement.

In the case of automobile gears when the duration of contact is small and tooth pressures are high, it is sometimes an advantage to make a modification of the correct theoretical shape and this is most readily performed by arranging a slight difference between

the pressure angle of the pinion and wheel teeth.

Having dealt briefly with some of the results of inaccuracies it is now proposed to consider how errors can be detected.

For the purpose of examination errors can be divided into three sections :

Error of concentricity.
,, ,, pitch and
,, ,, profile.

Error of concentricity means that the pitch line of the gear is not concentric with the bore from which the gear will ultimately run. In certain cases where gears are required to run with minimum backlash this eccentricity may cause jamming at one point and excessive backlash at the opposite point. Eccentricity causes fluctuation in the angular velocity which, if coinciding with a natural frequency of vibration, will cause considerable variations in tooth loading. The errors can best be detected by engaging the gear to be tested with a master gear of known accuracy on mandrills at the required centre distance.

The bed of the machine for making this test carries two slides, one mounted on balls and located in a definite zero position by means of a taper plunger pin, the other carried on guides and adjustable by means of a screw.

The gear to be tested and master gear are now rotated by hand to check for jamming, and feeler gauges can be inserted between engaging tooth profiles to examine for backlash. The following figures give a fair average for backlash :

Centre distance	4″	12″	20″	40″	60″
Backlash between teeth ...	4/1000″	6/1000″	8/1000″	12/1000″	14/1000″.

To test for concentricity, the locking plunger is withdrawn when a small spring presses the two gears tightly into engagement. If the gears are now rotated, any errors will cause a variation in the centre distance which can be observed on the dial indicator.

Pitch and profile errors will cause a sudden jump of the pointer from its normal position while eccentricity causes a gradual swing to a maximum and minimum during one revolution of the gear being tested.

For testing stem gears a vertical support with one fixed and one adjustable centre, is bolted to the fixed slide.

Before describing the instruments for measuring pitch errors let us consider exactly what to measure or compare.

When two gears are running together, only one set of profiles, either the driving or coasting, can be in use at any moment. This suggests, therefore, that each set of profiles should be measured and considered quite separately.

Further, it will obviously be advantageous if such measurements

can be made between the points on adjacent profiles which are in contact at the same moment.

Let us for a moment think of the generation of an involute. This curve is described by a point on the circumference of a circle as it is rolled along a straight line. A simple method of describing an involute is by means of a pencil attached to a piece of cotton or string wound round a cylinder. The cotton is kept taut as it is unwound and the pencil traces out the involute curve corresponding to the diameter of the cylinder. Imagine now, three pencils tied at equal distances along the cotton, then, as it is unwound three involutes will be described, and the distance between them measured along lines tangential to the base circle will always be equal. This is shown in Fig. 12.

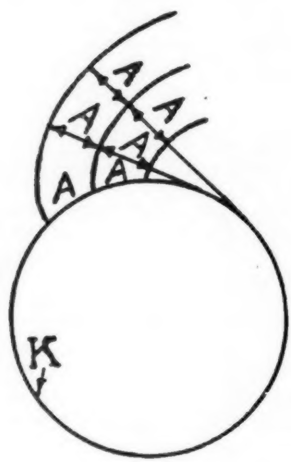

FIG. 12.

These distances " A " are all equal and as the cotton was kept taut when they were described they are the shortest distances between the profiles. The line of action is tangential to the base circle and consequently, the distance between profiles measured along it must also be equal to " A."

To compare the shortest distance between adjacent teeth around a gear, an instrument has been developed carrying a location face mounted on a slide and fitted with a finger to ensure correct seating. A lipped pointer is also provided, pivotted in the centre, free at one end and connected at the other end through a suitable lever system to a dial indicator, graduated in 1/10,000″.

This distance between the lip of the pointer and the flat location face is now adjusted until it approximates the distance between two adjacent profiles. The location face is then placed against one of the profiles and the instrument rolled on the tooth so that the lipped pointer travels down the adjacent profile. As the lip of the pointer moves downwards the distance becomes shorter and the pointer on the dial indicator moves steadily in one direction. On reaching the line of action, i.e. the shortest distance, the pointer remains stationary and then gradually moves back to its original position as the lipped pointer moves further down the tooth. The maximum reading on the dial indicator is observed and compared with readings taken on the remaining teeth around the gear. These comparisons are made on one set of profiles only, and are quite independent of the accuracy or otherwise of opposite profiles which are checked, if necessary, in a similar manner by turning either the instrument or the gear over. This gives a very good indication of how the teeth will take up their load as they move into engagement.

As all contacts take place along the line of action the distance between adjacent driving profiles of the pinion and driven profiles of the wheel must be equal when measured along this line and consequently the readings taken for the pinion and wheel should be similar with the same setting of the instrument. Any difference between the average reading for the pinion teeth as compared with the average reading for the wheel teeth indicates a difference in the pressure angle as the distance between profiles along the line of action is also the base circle pitch as shown in Fig. 12.

It is important to have some definite standard to which reference can be made from time to time to ensure that a constant pressure angle is being maintained. Gauges are made for use with this instrument, the distance between two vertical faces being equal to the base circle pitch corresponding to any particular diametral or circular pitch and pressure angle. This pitch remains constant for any number of teeth.

The instrument is set roughly to the required pitch and the gauge is then inserted between the location face and lipped pointer. The position of the pointer on the dial indicator is noted and all gears of that pitch and pressure angle should give the same reading.

Errors in profile are just as important as errors in pitch. An instrument for detecting such inaccuracies consists of a base carrying two slides at right angles to each other. One carries a mandril mounted on ball bearings on which the gear to be tested is fitted and also a disc ground exactly to the base circle diameter of the gear. This slide is provided with a screw for adjustment and a compression spring to keep a uniform pressure between the base circle disc and a straight edge carried on the other slide. Immediately above this straight edge a finger is located connected through

a system giving a magnification about 250 times to a recording device.

If the base circle is now pressed against the straight edge and the slide carrying the straight edge moved along its guide the disc will be rotated and any point on the circumference will trace out the involute corresponding to its diameter.

Assuming that the profile of the gear being tested has this theoretically correct shape there will be no movement of the finger which is lightly pressing against it and the recording pen will describe a straight line parallel to a zero line.

Where the tooth profile is a good involute but generated from some base circle other than the correct one the recorded line will be straight but at an angle to the zero line while a ragged line indicates a bad involute. These diagrams provide a permanent record of gears and also enable the engaging profiles of pinion and wheel to be compared for similarity of pressure angle and, while it does not matter, within fairly wide limits, what pressure angle is used, it is of vital importance that this angle is the same for both.

The methods of measuring, just described, have been developed to enable, as far as possible, a uniform quality to be maintained.

The demand for higher speeds, heavier tooth loads, and increased efficiency etc., have made it necessary to isolate, as far as possible, and to consider separately, the different types of errors as it is only in this way that definite progress is made towards reducing or eliminating them.

The following are a few examples of spur gear units having case-hardened and ground pinion or pinion and wheel :—

Gear for transmitting 6,400 h.p. from turbine running at 4,500 to generator at 1,500 r.p.m. Pitch line speed 12,200 ft. per minute. Tooth loading 790 lbs. per inch.

Gear for transmitting 13,600 h.p., increasing from motor at 750 r.p.m. to pump at 1,000 r.p.m. Pitch line speed 11,000 ft. per minute. Tooth loading 1,280 lbs. per inch width.

Turbo locomotive drive 2,440 h.p. reducing from 7,800/420 r.p.m.

Colliery gear 640/850 h.p. reducing 390/42 r.p.m.

Turbo exhauster 183 h.p. increasing 810/3,500 r.p.m.

While it is necessary to have accurate methods for inspecting gears, it is even more necessary to have machines capable of securing the required accuracy. The gear grinding and gear cutting machines developed by the Maag Company have several features which will probably be of interest.

These machines are made in various sizes but as they are similar in principle it is proposed to describe only the high speed model of each type.

The Maag Gear Planing Machine is of the vertical type, special care having been taken in the designing stages to obtain that rigidity so essential for good output and accuracy. It is of single

pulley drive through a metal to metal cone clutch to the gear box, the final drive being by fixed gear reduction to the crank disc driving the cutter slide. The gear blank is arranged on a horizontal table which is given a linear and rotary motion to secure the effect of a disc having the pitch circle diameter of the gear rolling along a straight line. These two motions are supplied by two sets of change wheels, one set coupled to the lead screw, the other set to a worm.

The rack type of cutter is used as being very strong and the straight profiles comparatively simple to produce accurately. This cutter is supported by a spring steel backing plate to take the thrust of the cut when the cutter teeth are worn thin. The cutter projects just the depth of the teeth from the ram, avoiding overhang and reducing the possibility of vibration. To ensure the cutter being parallel to the work table an instrument is clamped to a scraped face on the cutter slide carrying a swinging arm with an adjustable finger to touch the tips of the cutter teeth. Screws are provided to adjust the cutter at either end to bring it parallel before clamping. The setting instrument carries a vernier scale from which can be read the distance from the face on the cutter slide to the tips of the cutter teeth. When this is added to the root radius of the gear the tooth depth can be set on a scale of the bed of the machine without any reference to the outside diameter of the gear blank.

To avoid errors arising from play, the leadscrew nut has an auxiliary nut engaging the same leadscrew as the main nut. This locking nut is made with an external thread so that it may be screwed into an enlarged portion of the main nut. Integral with the locking nut is a toothed segment which engages with a similar segment keyed on to a spindle parallel to the leadscrew. A slight relative rotation between the nut and the locking nut has the effect of taking up any play or wear between the threads of the nut and the leadscrew. This second shaft is oscillated frictionally and arranged to move slightly in advance of the leadscrew thus taking up all play no matter in which direction the table is moving. To take up backlash in the generating mechanism, the work is brought back a little more than one pitch during the reversing motion and afterwards fed forward slightly before the first cut is taken. On the high speed machine, the cutter slide is balanced by a band passing over a cam to a spring in tension. This cam is so shaped that the tension in the spring is fairly uniform irrespective of the position of the cutter slide. On the larger machines the cutter slides are balanced by a weight.

The machine developed by the Maag Company for grinding gears operates on the generating principle, the involute curve being obtained from a pitch block and two pairs of steel bands.

From the beginning it was realised that to obtain the degree of

uniformity necessary for interchangeability both sides of the tooth should be ground at the same setting. This was particularly so where large series of gears had to be dealt with such as in motor car manufacture. These two grinding wheels were set at an angle to represent one tooth of a rack cutter and were mounted on separate slides so that they could be moved together and apart to give any pitch. Flat grinding wheels were tried on the first machine, but it was found impossible to keep the grinding faces flat and in their correct positions due to diamond wear.

It is well-known by anyone acquainted with cylindrical grinding that to clean a wheel up properly, the diamond must present a sharp corner to the grinding wheel and also that this corner wears very quickly so that the operator has to turn the diamond holder frequently to obtain fresh corners.

As the location of the wheel face and, consequently, the accuracy of the finished product depended on the trueing diamond, the difficulties of maintaining the grinding plane in its correct position to 0.0001″ can be imagined. They were, in fact, so great that a way round this difficulty had to be sought. The solution was found by using saucer-shaped wheels cutting only on their outside edges. Fig. 22 shows the type of grinding wheel now used. The cutting edge, as it spins round, forms a perfect plane to the gear which is rolled against it and fed past and, moreover, still maintains a perfect plane even when the edge wears.

With the introduction of this shape of wheel, it was realised that some form of automatic compensation was necessary to take up the wheel wear. The automatic compensator used in 1914 proved so successful that the same principle is used to-day, the only alternation being constructional ones to facilitate manufacture. The type at present used consisting of a flat diamond about $\frac{1}{4}$″ across its face let into a feeler lever carried between two hardened and lapped steel points. At the other end of this lever is a platinum contact while about the centre an arm is fitted to the lever carrying a small roller pressed against a cam by a light spring. The cam which is circular with a single recess is driven by a small belt from the grinding motor and revolves continually, making one revolution every three or four seconds. When the recess in the cam comes opposite the roller, the feeler lever is pulled back by the tension of the spring and one of two things takes place. Either the diamond rests lightly against the edge of the grinding wheel for a fraction of a second and is then withdrawn by the cam pushing the roller forward or, if the grinding wheel is not in its position the platinum point makes contact with another insulated point opposite and closes a circuit for a solenoid. This releases a clutch which rotates a shaft for one revolution before being uncoupled by a fixed cam. The shaft has at its opposite end a pawl which feeds round a ratchet wheel coupled to the grinding wheel slide. The

movement of the ratchet wheel can be set to vary according to the amount of stock to be removed, one tooth representing 0.00004".

Before commencing work, the grinding wheel is brought into its position by quick adjustment and thereafter maintained in that position automatically. This correction is of particular importance during the roughing operation as the whole success of gear grinding depends on being able to remove the distortion quickly and leave the gear accurate in pitch and profile ready for a light finishing cut.

The provision for taking up wheel wear makes possible the use of free cutting grinding wheels where the bond is soft enough to

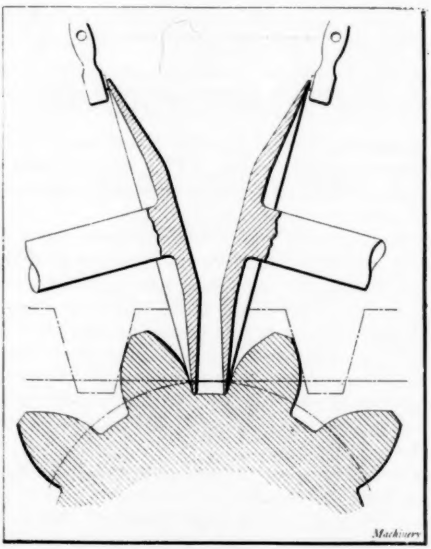

FIG. 22.

allow the grit to fall away immediately it becomes dull and to present fresh sharp cutting grit.

The machine has a traverse for grinding gears up to $5\frac{1}{2}$" face width dividing being set to take place at either or both ends of the gear as required.

Immediately the gear is clear of the grinding wheels, dividing

takes place and the traverse motion is then reversed and the gear fed back presenting a fresh tooth to the grinding wheels, and so on.

Although the angle of each of the grinding wheels is separately adjustable, this adjustment is only intended to enable such small corrections to be made to the pressure angle as may be necessary to secure correct meshing.

The grinding wheels are set normally at 15°, corrections for other pressure angles being made on the pitch block. It was explained at the beginning of this paper that precisely the same involute can be generated with cutters having any pressure angle providing the generation takes place on a circle corresponding to the particular angle used. Thus, if a gear is to run with a 20° angle the teeth may be cut with a 20° cutter generating on the pitch line on say a 15° cutter generating a diameter equal to :

$$\frac{\text{pitch circle} \times \text{Cos. } 20^\circ}{\text{Cos. } 15^\circ.}$$

This system is adopted to avoid the necessity of swinging the grinding wheels to different angles and very much simplifies the setting up.

For equal distribution of stock, the pitch block tensioning arms are supported by a central column. This column has a short thread carrying a hand nut and as it is moved to the right or left, rotates the pitch block and gear.

Tooth spacing is by means of a dividing plate having teeth on its periphery case-hardened and afterwards ground to a high degree of accuracy. Location is by hardened and ground cylindrical bolt in the tapered spaces between the teeth of the dividing plate.

A toothed index disc on the front of the machine is fitted with two levers which may be set to change over from the roughing to the finishing feed and to stop the machine when the gear is finished.

On the larger machines, taking gears up to 48″ face width, the masses are too heavy for automatic dividing and this operation has to be performed by hand.

An attempt has been made in this paper to indicate the lines on which spur gearing is developing.

Prior to the introduction of gear grinding the inaccuracies in spur gears were too large for them to be used successfully for high speed work or for full advantage to be taken of the increased durability of hardened profiles. Improvements in gear grinding machines and in measuring instruments have, however, cleared away many of the complexities which the machining operation formerly possessed and there is now no reason why users of ground gears should not produce them in their own works.

OPENING OF THE LUTON, BEDFORD AND DISTRICT SECTION.

Inaugural Meeting, Lecture Room, Public Library, Luton, 11th December, 1929.

The following report of the proceedings of the Inaugural Meeting of the Luton, Bedford and District Section is reproduced from that published by the *Luton News* of 12th December, 1929.

HONOUR FOR LUTON.

CENTRE FOR A NEW ORGANISATION.

Last Night's Inaugural Meeting.

Last night, the lecture room of Luton Public Library was crowded with engineers for the inaugural meeting of Luton, Bedford and District Section of the Institution of Production Engineers. There were representatives of Davis Gas Stove Co., Ltd., Borough Engineering Works, Vauxhall Motors, Ltd., G. F. Farr, Luton Engineering Pattern Co., Adamant Engineering Co., L. Weeks, Ltd., Chas. Wilson and Sons, G. Kent, Ltd., Cundell Folding Machine Co., Shell Mex, General Motors, Skefko Ball Bearing Co., Ltd., T. Balmforth and Co., F. H. Eve, Ltd., and Commer Cars, Ltd., Luton; Heald Machine Co., British Tabulating Co., Foster Instrument Co., Kryn and Lahy, Letchworth; Allen and Sons and Motor, Rail and Tramcar Co., Bedford; Bagshawe and Co., Ltd., Dunstable; and also visitors from Welwyn and St. Albans.

Mr. R. W. Bedford, Works Manager, Messrs. G. Kent, Ltd., presided, and said it was most gratifying to the committee to see such a large company at the opening meeting of the Luton section of an institution destined to become an important and influential organisation associated with the engineering industry. The President of the Institution was Mr. Tom Thornycroft and among the vice-presidents was Mr. A. J. Hancock, who was President of the London Section. A message had been received from Sir Alfred Herbert, K.B.E., a Past President, wishing the new section success.

A GREAT COMPLIMENT.

It was a great compliment that Luton had been selected as the centre for this new section. It stamped the town as an important engineering centre with a great future. Almost every branch of engineering was represented in the town, in Bedford and in other parts of the area, so that the Section would undoubtedly serve a useful purpose in bringing together responsible men engaged in production to discuss problems and suggest solutions.

Mr. R. H Hutchinson, Past President of the Institution and production engineer of Messrs. D. Napier and Son, Ltd., spoke at some length on the objects of the Institution and described production engineering as the biggest branch of engineering in this country. There were great opportunities for production engineers, and it would not be long before the Institution held an official position in the technical education world. To achieve the end for which they aimed they must have weight and power which could only be attained through a big membership.

MANUFACTURING METHODS.

Mr. J. H. Garnet, A.M.I.Mech.E., Associate Editor of "Machinery," read a paper on "Progress in manufacturing method," illustrated by a number of interesting slides showing methods adopted in works in different parts of the country. The lecturer said British manufacturers could give as good an account of themselves as manufacturers in any part of the world, and considered

that the individuality put into British-made articles was preferred to mass production goods. Particular reference was made to methods adopted in the cutlery trade in the manufacture of table knives, to locks, heavy and hollow forgings, and huge boiler castings, and the use of fabricated steel for the last, of which the speaker prophesied a big future. If we were to succeed in capturing the markets hitherto regarded as our prerogative, a line of quality must be taken. It was in the direction of quality that our salvation lay, plus the judicial application of mechanical processes.

The lecturer was thanked by Mr. Carlton Smith and Mr. F. Siddall, and the Chairman and Mr. Hutchinson and Mr. Richard Hazleton, General Secretary of the Institution, were thanked by Mr. F. Southall and Mr. Ronald.

The Secretary of the Section is Mr. F. W. M. Lee, "Hillfoot," Crescent-rise, Luton.

ANNUAL REPORT FOR 1929-30.

To be Presented by the Council to the Annual General Meeting, London, 3rd December, 1930.

Membership.

The membership at the end of July, 1930, was 446, made up as follows :—

Ordinary Members	247
Associate Members	155
Graduates	16
Associates	14
Affiliates	12
Hon. Members	2
	446

Two hundred-and-one new members were elected during the year. Of these twenty-one elected late in the year had not completed membership at the end of June, though they have since done so. The actual number of new members who completed membership during the year was, therefore, 180. In addition, there are eleven Affiliated Firms.

A little over half the increase for the year is attributable to the two new Sections opened at Luton and Manchester.

The membership of the various Sections was made up as follows :—

Birmingham Section	123
Coventry Section	62
London	153
(103 London area).	
Luton, Bedford and District	38
Manchester	70
	446

Incorporation.

An application has been made to the Board of Trade for the Incorporation of the Institution under the Companies Act, 1929. If granted, as expected, this will give the Institution the legal status it has hitherto lacked and will constitute full protection for members in the matter of financial liability.

Memorandum and Articles of Association.

In view of the application for Incorporation, it became necessary to draw up a Memorandum of Association and to revise the

Articles. The opportunity was taken to make certain alterations in and additions to the Articles which experience showed to be desirable.

Affiliation of Engineering Societies.

A scheme for the affiliation of Engineering Associations has been completed during the year, and when meetings are resumed in the autumn it is hoped that the scheme will be the means of linking up certain of these organisations with the Institution.

Syllabus for Session 1930-31.

The Council congratulates Local Section Committees on the programmes drawn up for the coming Session. The ballot of members on the question of subjects for discussion was a help in several cases. A new departure which it is intended to maintain annually is the selection of one subject each year for consideration by all Sections. The subject selected for this purpose during the 1930-31 Session is: "Payment by Results." The Institution should be in a position, as a result, to issue a useful report on this subject.

Strengthening Membership.

A young Institution such as ours, if it is to progress, must rely mainly on the activity and enthusiasm of its members. The surest sign of a live organisation is a good record of attendance at meetings, which, in turn, stimulates good discussion. Each member on joining the Institution signed an "Engagement" which included this undertaking: "That I will advance the objects of the Institution as far as shall be in my power, and will attend the meetings thereof as often as I conveniently can." Members interested enough to attend meetings regularly will soon be interested enough to bring with them potential new members of the right calibre, and to take a pride in helping to strengthen the membership.

Finance and Organisation.

From its inception up to within six weeks of the beginning of the year under review, the Institution had no paid staff or office of its own. It was felt that the time had come to make a change in this respect. In May, 1929, Mr. Hazleton was appointed General Secretary, and a small office was opened at Rupert Street. This step has already been justified. Not only has all expenditure for the year been met out of revenue, but it has been possible to increase our Deposit Account at the Bank by £150. Though the staff still comprises only the General Secretary and one assistant, larger offices (including general office, Council room and library) were taken in Great James Street in June, 1930. These should meet our needs for some time to come.

Meetings.

Thirty-two meetings were held during last Session in five centres. The number of lectures and addresses given by members was fifteen. Attendance at meetings was in most cases very gratifying, the average attendance at Birmingham Section meetings, for instance, being between 100 and 150. The most popular Paper given during the Session was "Moulds for Synthetic and Other Compounds," by Mr. H. T. Richardson, Associate Member, Coventry Section. Mr. Richardson has since been made a full Member. Birmingham Section has made a practice of having two or three Informal Discussions during the Session. The Informal Discussions are usually confined to members only. As the proportion of visitors at meetings is a large one—and rightly so at this stage of the Institution's development—the practice of giving members the exclusive advantage of certain discussions has much to commend it.

Publications.

The Institution has now three regular publications, *The Journal*, which is published monthly and is a record of the Proceedings; *The Bulletin*, published bi-monthly, composed mainly of notes of interest to members, and *The Circular of Information*, also bi-monthly, but alternating with *The Bulletin*. Publication of the Proceedings of last Session is now well advanced and should be completed early in 1931. *The Journal* is available to non-members, and has a steadily growing circulation.

Local Sections.

Two new Sections, one at Luton, the other at Manchester, were opened during the year, and have made a successful start. London was also formally constituted into a Section.

Examination Scheme.

As the result of conferences with the representatives of the organisations of Principals and Teachers in Technical Institutions a joint Committee was set up during the year for the purpose of preparing an agreed scheme on a national basis for an Institution Examination for Graduate Membership. As a result the scheme is now in operation and the first Examination will be held at Easter, 1931.

Visits and Other Functions.

A number of visits to engineering works were arranged by Sections during the year, and in September an official visit for the whole membership of the Institution was paid to the Shipping and Engineering Exhibition, Olympia. Sir Alfred Herbert, President of the Institution, attended and spoke on that occasion, and also presided at the Annual Dinner in London in October, when

Mr. Tom Thornycroft, President-Elect, Sir Herbert Austin, and Mr. Evans, President of the Association of Teachers in Technical Institutions, were among the chief guests. Two of the Sections (London and Luton) have started the practice of combining their Annual Section Meetings with a Supper. The innovation has proved a welcome one.

President's Prize.

Sir Alfred Herbert, president, very kindly offered a prize of £10 for the best Paper submitted by members on the subject of "Electrical Drive of Machine Tools." The prize was won by Mr. D. Nicholson, Member, whose Paper was published in *The Journal* in July, 1930. Mr. Tom Thornycroft, our distinguished President for the past year, has offered a similar Prize.

Presentation Clock.

The Council has to thank the London Section Committee for presenting a handsome clock, of English manufacture, to the Institution, to mark the opening of the new offices at Great James Street. The clock now stands in the Council Room.

Library.

Space is available at the new offices in Great James Street for a small Library, and already a nucleus of books, periodicals, *Proceedings* of other engineering institutions, etc., has been formed.

Appointments Bureau.

The Bureau has provided a valuable service during the year for members seeking new posts, and has been availed of by many firms as a means of securing engineering production officers. It is felt, however, that members in a position effectively to help the Bureau by the notification of vacancies do not yet make the fullest use of the facilities it offers, and it is hoped that next year there will be an improvement to record in this respect. Up to the time of preparing this Report, the very serious industrial depression does not appear to have led to unemployment among members, though in the last quarter of the year vacancies notified have been less in number than usual.

INSTITUTION OF PRODUCTION ENGINEERS.

BALANCE SHEET AS AT 30 JUNE, 1930.

LIABILITIES.

	£ s. d.	£ s. d.
Sundry Creditors		62 16 2
Subscriptions received in advance		4 10 0
INCOME AND EXPENDITURE ACCOUNT :		
Balance at 1 July, 1929 ...	525 4 5	
Add Excess of Income over Expenditure	178 6 0	
		703 10 5

ASSETS.

	£ s. d.	£ s. d.
FURNITURE AND FITTINGS.		
Balance at 1 July, 1929	26 15 0	
Additions during year ...	63 5 8	
	90 0 8	
Less depreciation	9 0 0	
		81 0 8
SUNDRY DEBTORS :		
Advertisers Accounts ...	55 12 11	
General	39 10 0	
Prepaid Charges	10 0 0	
		105 2 11
CASH :		
Deposit Account	500 0 0	
Current Account	43 7 5	
In hand	4 7 3	
		547 14 8

			Cash at Sections :							
			Current Accounts	35	6	0				
			In hand	1	12	4				
								36	18	4
£770	16	7						£770	16	7

Auditor's Report.—We have audited the above Balance Sheet, dated the 30th June, 1930, and we have obtained all the information and explanations we have required. In our opinion such Balance Sheet is properly drawn up so as to exhibit a true and correct view of the state of the Institution's affairs according to the best of our information and the explanations given us and as shown by the books of the Institution.

(Signed) C. H. APPLEBY. and CO.,
Auditors.
Chartered Accountants.

3, Newman's Court, Cornhill, London, E.C.3.
10 September, 1930.

(Signed) John A. Hannay, *Chairman of Council.*

(Signed) Robert Hutchinson, *Chairman, Finance Committee.*

(Signed) Richard Hazleton, *General Secretary.*

INCOME AND EXPENDITURE ACCOUNT FOR THE YEAR ENDED 30 JUNE, 1930.

Dr.

	£	s.	d.	£	s.	d.
To Administration Expenses :						
Salaries	510	13	9			
Rent	52	12	0			
Printing and Stationery	86	3	1			
Lighting, Heating, and Cleaning	15	15	11			
Telephone	15	10	9			
Postages	43	12	6			
Travelling and Hotel Expenses	26	12	7			
Audit Fee	5	5	0			
Bank Charges	1	13	7			
Insurance	1	6	9			
Expenses of Meetings...	17	10	7			
Depreciation of Office Furniture	9	0	0			
Removal Expenses ...	10	0	0			

Cr.

	£	s.	d.
By Subscriptions	1,062	0	0
,, Interest on Deposit Account ...	11	1	9
,, Bulletin Advertisements	11	6	11
,, *Journal* :			
Receipts from Sales and Advertisements, etc.	239	0	8

	Sundries	11	16	0									
					807	12	6						
,,	SECTION EXPENSES :												
	London	25	2	8									
	Birmingham...	41	6	1									
	Coventry	46	18	3									
	Manchester	32	14	10									
	Luton	11	1	9									
					157	3	7						
,,	ANNUAL DINNER :												
	Expenditure...	52	12	0									
	Less Receipts	37	14	0									
					14	18	0						
,,	*Journal* EXPENSES :												
	Printing, Postage, and Stationery				165	9	3						
,,	Balance, being excess of Income over Expenditure				178	6	0						
					£1,323	9	4					£1,323	9	4